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HEAT-TRANSFER TESTS OF SEVERAL ENGINE COOLANTS

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SUMMARY

The relative cooling performance of water, commercial Prestone (ethylene glycol plus additives), and several ethylene-glycol and diethylene-glycol derivatives was obtained from tests conducted with a small electrically heated heat exchanger. The comparison of the cooling performances in this report covers only the turbulent-flow region under conditions where the effects of film boiling and free convection are negligible. Because all of the pertinent physical properties of the organic coolants were not available, the heat-transfer coefficients obtained were plotted as a function of average coolant temperature and weight rate of coolant flow.

The relative cooling performance of water is much higher than that of any of the organic coolants tested, being 4.4 times greater than that of Prestone at 200° F. Four of the coolants tested had cooling performances at 200° F ranging from 1.09 to 1.23 times that of Prestone.

INTRODUCTION

The present extensive use of liquid-cooled aircraft engines has restimulated interest in the search for a more satisfactory high-temperature coolant than ethylene glycol. The requirements imposed on an engine coolant are varied and severe. It must be capable of withstanding high temperatures (250° F or higher) in order that radiator sizes may be kept to a minimum. At the same time the properties of the coolant should be such that a high rate of heat transfer is obtained between the cylinder walls and the coolant in order to maintain low cylinder-wall temperatures. In addition, the coolant should have a high flash point for safety considerations and should be stable and chemically inert toward the materials used in the cooling system.

Ethylene glycol has been widely used as an aircraft-engine coolant in the United States because it has a high boiling point that permits operation at coolant temperatures of approximately 250° F with attendant small radiator sizes. The use of aqueous solutions of ethylene glycol results in increased heat-transfer performance compared with that of straight ethylene glycol but requires that the cooling system be pressurized if the same coolant temperature is to be obtained.

Results of unpublished tests conducted by the Ethyl Corporation with ethylene glycol, diethylene glycol, triethylene glycol, Cellosolve, butyl Cellosolve, and aqueous solutions of these fluids in a supercharged CFR engine showed that both Cellosolve and butyl Cellosolve gave better cooling than did ethylene glycol. The conclusion was reached that the boiling point of the chief constituent of the coolant was the most important property affecting the heat transfer between the cylinder wall and the coolant. The tests were conducted under conditions favorable to boiling; that is, a very low coolant flow rate combined with a high coolant temperature.

Frank (reference 1) showed that the temperature difference between the coolant and the cylinder wall was much larger when ethylene glycol rather than water was used as a coolant. Wood (reference 2) observed a similar condition and attempted to explain the phenomenon in terms of the latent heat of vaporization and the specific heats of the two fluids.

A search of the literature was made in an effort to obtain a list of fluids that would be suitable for test work. Most fluids failed to meet one or more of the previously mentioned requirements of a high-temperature coolant. The list was finally composed of seven organic liquids, all ethylene-glycol or diethylene-glycol derivatives.

This preliminary study was hampered, in many cases, by lack of knowledge of the physical properties of the fluids, especially thermal conductivity. Estimates of the thermal conductivity were made for promising fluids by means of an empirical equation derived by Smith in reference 3. It should be noted that, if thermal conductivity, viscosity, and specific heat are known for a fluid, its heat-transfer performance in the turbulent-flow range can be directly predicted. For most of the fluids tested some of these properties were not available; hence experimental work was necessary to determine the heat-transfer coefficients.

A small heat-transfer unit rather than an engine was selected for the experimental work inasmuch as heat-transfer data from engines are apt to be erratic owing to the difficulty of controlling

the many variables present. The heat-transfer unit was designed to
simulate engine conditions in that the heat input per unit area
corresponded to the high rates obtained in liquid-cooled engines.

A forced-convection, turbulent-flow basis was chosen as that
likely to correlate heat-transfer data obtained from liquid-cooled
engines because coolant flow rates are high and the flow is turbu-
lent in the coolant passages. Preliminary heat-transfer data
obtained from a liquid-cooled cylinder showed that this choice was
justified although some evidence indicated that the heat-transfer
processes on the liquid side are complicated by other modes of heat
transfer, such as film boiling.

Because no data found in the literature presented the compara-
tive characteristics of engine coolants in terms of heat-transfer
theory, experimental tests were conducted to form a rational method
for the comparison of liquid coolants on a heat-transfer basis and
to compare the heat-transfer properties of several promising organic
coolants. The results of the experimental tests and an analysis of
the results are presented herein.

SYMBOLS

c_p specific heat at constant pressure, (Btu)/(lb)(^{o}F)

Pr   Prandtl number (Pr $= c_p\mu/k$)

D   equivalent diameter $(D = D_1 - D_0)$, (ft)

D_1 inside diameter of Micarta jacket, (ft)

D_0 outside diameter of monel-metal tube, (ft)

h   surface heat-transfer coefficient, (Btu)/(sec)(sq ft)(^{o}F)

Nu   Nusselt number (Nu $= hD/k$)

h/w^n cooling-performance factor

k   thermal conductivity, (Btu)/(sec)(sq ft)(^{o}F/ft)

T   average coolant temperature, (^{o}F)

V   fluid velocity, (ft)/(sec)

Re   Reynolds number, (Re $= VD\rho/\mu$)

W   weight rate of flow, (lb)/(sec)

μ absolute viscosity, (lb)/(sec)(ft)

ρ fluid density, (lb)/(cu ft)

n, m   exponents, experimentally determined

ANALYSIS

The heat-transfer coefficient is, for forced-convection turbulent flow, a function of the weight rate of flow and the various properties of the coolant. This relation may be derived by dimensional analysis and is given symbolically by expressing the Nusselt number as a power function of the Reynolds and Prandtl numbers:

$$\frac{hD}{k} = \phi \left(\frac{VD\rho}{\mu}\right)^n \left(\frac{c_p\mu}{k}\right)^m \tag{1}$$

The numerical values of the constant ϕ and the exponents n and m may vary according to whether the flow is upward, downward, or horizontal and also according to whether the fluid is being heated or cooled. For one given flow condition, however, ϕ and the exponents n and m are constant regardless of the type of liquid being tested.

For the purpose of the present tests, equation (1) may be written

$$\frac{h}{W^n} = \phi' \; k \left(\frac{1}{\mu}\right) \left(\frac{c_p\mu}{k}\right)^m \tag{2}$$

where ϕ' is a constant including the characteristic diameter D. The relation given in equations (1) and (2) is of great value in predicting heat-transfer coefficients if the fluid properties included in the equation are known. Because c_p, μ, and k are functions only of temperature, equation (2) may be written for one liquid as

$$\frac{h}{W^n} = \phi'' \; f \; (T) \tag{3}$$

Equation (3) is convenient for the purpose of rating coolants when their properties are unknown. When h/W^n is plotted against T, a separate curve is obtained for each fluid. The fluid giving

the higher heat-transfer coefficient for a given flow rate will produce the larger value of the left-hand member of equation (3), and the curve for the better coolant will therefore lie above the curve for a less efficient coolant. Consequently, the parameter h/W^n is a measure of the cooling performance given by any fluid in a purely forced-convection, turbulent flow, heat-transfer process and will hereinafter be termed the "cooling-performance factor."

If equation (1) is transformed into the form of equation (3), it is necessary only to plot h against W at a constant coolant temperature in order to determine the value of the exponent n.

COOLANTS

A summary of the coolants tested and their physical properties, obtained from reference 4, are given in the following table:

Coolant	Boiling point at 760 mm ($^\circ$F)	Flash point ($^\circ$F)	Specific gravity at 68°/68° F	Viscosity at 68° F (centipoises)
Prestone	[a]387	[a]240	[a]1.1155	[a]21.00
Water	212	-----	1.0000	1.01
Diethylene glycol	473	290	1.1184	37.70
Carbitol	395	210	.9898	4.50
Butyl Carbitol	447	240	.9555	6.35
Methyl Carbitol	380	200	1.0354	3.91
Diethyl Carbitol	370	180	.9094	1.39
Benzyl Cellosolve	493	265	1.0700	------------

[a]These values are for pure ethylene glycol and are therefore approximate for the Prestone used in the tests.

APPARATUS

The heat-transfer unit used for the experimental tests is shown in figure 1 and consisted of a monel-metal tube of 1/2-inch outside diameter and $15\frac{1}{2}$-inch length. The wall thickness was 1/32 inch. Hollow copper terminals were silver-soldered on the ends of the tube making the test section 14 inches long. Ten chromel-constantan thermocouples, spaced along the tube in two diametrically opposite rows, were soldered flush with the surface of the tube with a high melting-point soft solder. The thermocouple wires were brought out through the inside of the tube and the terminals. Because the last

point of contact of the two wires forming the thermocouple hot
junctions was at the inside of the tube, the temperature measured
was that of the inside surface of the tube. The tube was surrounded
by a Micarta jacket of 3/4-inch inside diameter that permitted the
liquid to flow vertically through an annular passage 1/8 inch wide.

The tube terminals were clamped to the secondary terminals of
the transformer. (See fig. 2.) The voltage across the primary
side of the transformer could be closely adjusted by means of a
saturable reactor-type voltage regulator. A wattmeter, two volt-
meters, two ammeters, a current transformer, and a potential trans-
former were used to measure the power, and hence the heat input, to
the tube.

The current transformer stepped down the tube current to a
value commensurate with the range of the current coil in the watt-
meter, and the potential transformer performed a similar function
by stepping up the low voltage drop across the monel-metal tube.
Because the transformation ratio of the potential transformer
varied slightly with load, the ratio was determined during each
run by voltmeters placed across the primary and secondary sides.
A correction was applied to the power measurements because of the
voltage drop in the potential leads. This voltage drop was calcu-
lated from the known resistance of the leads and the current in
the leads, which was determined by an ammeter. The effect of phase-
angle shift on the potential transformer was checked with an oscil-
loscope and was found to be negligible.

The inlet and outlet temperatures of the coolant were measured
by separate thermopiles consisting of four chromel-constantan
thermocouples connected in series and spaced across the inlet and
outlet flow passages. These two thermopiles were differentially
connected at the cold-junction box so that their voltages were
opposed. Thus it was possible to read four times the average elec-
tromotive force at both the inlet and outlet sections and four
times the difference in average electromotive force between the
inlet and outlet sections.

Figure 3 shows the arrangement of the apparatus. The coolant
flow rate was measured with a flowmeter of the rotameter type that
was calibrated in place for each liquid by means of a weighing tank
and scale. The flow was obtained from a centrifugal pump that pro-
duced a maximum flow of approximately 8 gallons per minute.

Mercury-in-glass thermometers located in wells at the inlet and
outlet sections permitted direct observation of the coolant tempera-
tures. The inlet-coolant temperature was controlled by an electri-
cal preheater and a water-cooled heat exchanger. The cooler could
be bypassed to permit increased flow rates.

TEST PROCEDURE

The experimental procedure consisted in adjusting the tube current to the desired value by means of the voltage regulator. The coolant temperature was set by means of the preheater and the cooler. The coolant flow was regulated by throttling valves in the outlet side of the pump and by the bypass valve on the cooler.

When equilibrium was attained, the electrical meters were read and thermocouple readings were taken with a portable potentiometer. The meters were again read at the completion of the run and the two values were averaged.

Two series of runs were made for each coolant. In the first series, the average coolant temperature was maintained constant and the weight flow was varied. In the second series, the weight flow was held approximately constant and the coolant temperature was varied. The first series was made for the purpose of obtaining the exponent n of the weight flow rate W, and the second series permitted the value of h/W^n to be plotted against temperature.

The maximum tube current was approximately 1100 amperes, which corresponds to a heat flux of 101,000 Btu per square foot per hour. In order to obtain an accurate value of the heat-transfer coefficient h, it was desirable that the power input to the tube be as high as possible. This maximum heat input was determined by varying the tube current and holding the coolant flow rate and coolant temperature constant. When the calculated value of the heat-transfer coefficient ceased to become independent of the tube current, this current was assumed to be the maximum that could be used without introducing additional heat transfer due to free convection or boiling. The tube current was then slightly decreased to be certain that the data would not be affected by the previously mentioned factors.

The values of heat input obtained by measuring the flow rate and the temperature rise of the coolant agreed with few exceptions to within 5 percent of the values obtained by the electrical-power measurements. In all cases, however, the heat-transfer coefficient was calculated by using the values of the electrical-heat input because this measurement was thought to be the more reliable of the two owing to the difficulty of accurately measuring the small temperature rise of the coolant.

A summary of the experimental data is presented in table I.

DISCUSSION

In order to plot the data in the form given by equation (3), the value of the exponent n was determined by plotting h against W for several coolants, as shown in figure 4, and by measuring the slope of the curves. From figure 4 the value 0.8 was selected as an average slope that most nearly represented the data for all the coolants. The result of plotting $h/W^{0.8}$ against T for the coolants tested is shown in figure 5. The relative cooling-performance factors at 200° F of all liquids tested when the factor obtained with Prestone is taken as unity are as follows:

Relative cooling-
performance factor,

$$\frac{(h/W^n)_{coolant}}{(h/W^n)_{Prestone}}$$

Prestone	1.00
Water	4.46
Diethylene glycol	.71
Carbitol	1.16
Butyl Carbitol	.96
Methyl Carbitol	1.23
Diethyl Carbitol	1.14
Benzyl Cellosolve	1.09

From an inspection of the values it is seen that of the organic coolants, diethyl Carbitol, methyl Carbitol, Carbitol, and benzyl Cellosolve are slightly more effective than Prestone for cooling in the forced-convection range; whereas diethylene glycol and butyl Carbitol are slightly less effective. Water, of course, is much more effective than any of the organic coolants.

Because the heat generation in the small plug of solder at the junction of the thermocouple wires was not the same as that in the tube wall proper and because the point of contact of the thermocouple wires nearest the potentiometer was at the inside surface of the tube, the wall temperatures listed in table I are not the true surface temperatures. The relative values of the cooling-performance factors of the various coolants tested should not, however, be greatly affected even though the true surface temperatures of the wall were not measured.

Operating characteristics of the heater tube were very satis-factory. Equilibrium conditions were quickly obtained when the

electrical input to the tube was changed. An automatic, liquid temperature controller would have been an improvement on the test setup to facilitate close control of coolant temperatures.

A preliminary study of the chlorinated hydrocarbons, some of which are noninflammable, was discontinued because the heat-transfer properties of this group were found to be very poor. In addition, most of these compounds tend to become unstable in the presence of air, hot metal, or both.

The results of this investigation were discouraging in that no high-boiling coolant was found that had markedly better heat-transfer properties than ethylene glycol. It is unlikely that any common organic fluid can be found that will approach the cooling performance given by water. An ideal solution to the cooling problem would lie in the discovery of a substance which, when added to water, would raise its boiling point at atmospheric pressure to 250° F or higher without altering its heat-transfer properties.

SUMMARY OF RESULTS

Tests with a small heat-transfer unit indicated that the cooling performance of water is greatly superior to all seven organic coolants tested.

The use of an electrically heated tube for the determination of heat-transfer coefficients was found to be practicable because:

(a) A high heat flux could be readily obtained.

(b) The determination of the heat input was simple, fast, and accurate.

(c) Equilibrium conditions could be quickly obtained.

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REFERENCES

1. Frank, Gerhardt W.: High-Temperature Liquid-Cooling. SAE Trans.,
     vol. XXV, no. 4, Oct. 1929, pp. 329-343.

2. Wood, H.: Liquid-Cooled Aero Engines. SAE Trans., vol. 39,
     no. 1, July 1936, pp. 267-286.

3. Smith, J. F. Downie: Thermal Conductivity of Liquids. Ind. and
     Eng. Chem., vol. 22, no. 11, Nov. 1, 1930, pp. 1246-1251.

4. Anon.: Fine Chemicals. Carbide and Carbon Chemicals Corp.
     (New York), 1939, pp. 26-29.

TABLE I - SUMMARY OF HEAT-EXCHANGER DATA

Coolant	Coolant flow, W (lb/min)	Coolant-in temperature (°F)	Power input to tube (watts)	Average tube temperature (°F)	Coolant-out temperature (°F)	Surface heat-transfer coefficient, h [Btu/(sec)(sq ft)(°F)]
Prestone	37.8	160.0	2090	291.0	165.0	0.102
	46.7	162.0	2093	271.0	166.0	.121
	52.6	158.0	2093	260.0	162.0	.130
	58.7	161.0	2088	253.0	165.0	.144
	30.4	202.0	2430	345.0	208.0	.107
	39.0	205.0	2422	318.0	209.0	.135
	47.9	204.0	2427	300.0	208.0	.160
	53.9	205.0	2427	290.0	209.0	.181
	60.0	204.0	2409	282.0	208.0	.196
	56.7	116.0	3670	308.0	123.0	.120
	54.2	130.0	3566	318.0	136.0	.120
	56.0	150.0	3660	312.0	156.0	.143
	56.0	159.0	4425	342.0	166.0	.153
	59.1	191.0	3645	313.0	194.0	.188
	60.0	222.0	4278	340.0	228.0	.231
	59.4	232.0	3645	330.0	237.0	.238
	59.8	242.0	3652	336.0	248.0	.249
Water	17.5	60.0	2696	158.0	71.0	0.180
	22.2	60.0	2720	146.0	68.0	.206
	27.0	60.0	2731	137.0	66.0	.229
	32.3	60.0	2760	129.0	65.0	.260
	37.6	60.0	2760	123.0	65.0	.282
	32.2	139.0	2527	182.0	144.0	.392
	37.3	141.0	2538	179.0	144.0	.438
	41.7	142.0	2558	177.0	144.0	.467
	43.2	142.0	2558	173.0	144.0	.529
	55.0	141.0	2561	171.0	144.0	.668
	60.5	142.0	2561	170.0	144.0	.589
	17.5	118.0	2773	191.3	127.0	.250
	17.5	154.0	2782	216.0	164.0	.303
	17.5	194.0	2746	254.1	204.0	.370
	37.8	120.0	2774	160.1	125.0	.458
	37.8	140.0	2765	175.3	144.0	.506
	37.8	154.0	2765	188.9	158.0	.522
	37.8	175.0	2749	208.4	180.0	.552
	37.8	203.0	2741	233.1	206.0	.595
	51.8	138.0	4137	183.5	143.0	.597
	51.8	155.0	4404	201.1	159.0	.620
	51.8	175.5	4354	219.2	182.0	.668
	57.9	116.5	4124	161.5	121.5	.603
	58.0	136.0	4181	180.0	140.0	.618
	56.9	147.0	1195	159.5	149.0	.646
	58.9	191.0	1651	207.9	193.0	.790
	57.9	188.0	4365	218.5	183.0	.819
	58.9	193.0	4185	229.1	197.0	.760
Diethylene glycol	52.0	102.0	2236	286.1	106.0	0.076
	57.5	115.0	2430	328.8	122.0	.072
	61.4	136.0	2477	299.6	139.0	.095
	60.5	160.0	2238	287.2	162.0	.110
	61.0	176.0	2425	301.1	180.0	.123
	62.0	199.0	2521	311.8	202.0	.141
	62.2	216.0	2494	319.1	220.0	.153
	62.5	231.0	2518	329.3	235.0	.162
Carbitol	45.0	100.0	2941	278.0	106.0	0.104
	50.9	102.0	2936	270.0	106.0	.110
	56.5	102.0	2945	261.0	106.0	.117
	62.5	99.0	2962	252.0	104.0	.123
	68.4	102.0	3012	244.0	105.0	.133
	46.9	147.0	2964	285.0	153.0	.136
	53.3	150.0	2976	275.0	154.0	.150
	59.2	149.0	2988	263.0	153.0	.166
	65.5	149.0	2968	254.0	153.0	.179
	71.4	148.0	3030	245.0	152.0	.198
	47.1	199.0	2942	302.0	205.0	.183
	54.0	200.0	2961	294.0	205.0	.202
	60.5	200.0	2941	282.0	204.0	.228
	66.3	197.0	2940	274.0	202.0	.244
	72.0	199.0	2940	269.0	202.0	.269

Coolant	Coolant flow, W (lb/min)	Coolant-in temperature (°F)	Power input to tube (watts)	Average tube temperature (°F)	Coolant-out temperature (°F)	Surface heat-transfer coefficient, h [Btu/(sec)(sq ft)(°F)]
Butyl Carbitol	55.2	193.0	4540	351.4	199.0	0.181
	61.4	195.0	4426	337.9	202.0	.197
	68.1	194.0	4415	333.6	199.0	.200
	53.5	99.0	4459	316.1	107.0	.130
	60.5	102.0	4483	312.4	110.0	.135
	67.5	107.0	4435	309.0	114.0	.139
	55.0	146.0	4465	326.8	153.0	.157
	61.2	146.0	4484	322.2	153.0	.161
	68.5	145.0	4536	316.2	151.0	.168
	64.0	137.0	4140	323.9	144.0	.140
	64.5	148.0	4163	322.9	154.0	.151
	64.6	152.0	4108	324.3	158.0	.151
	64.7	178.0	4178	329.9	174.0	.168
Methyl Carbitol	52.6	99.0	3688	278.7	106.0	0.124
	59.2	99.0	3698	271.6	105.0	.128
	65.1	98.0	3664	259.7	104.0	.134
	72.5	102.0	3672	247.3	107.0	.160
	48.9	146.0	3539	286.7	153.0	.160
	55.5	145.0	3611	279.2	150.0	.170
	61.1	146.0	3620	269.1	152.0	.187
	67.9	147.0	3714	262.4	152.0	.204
	74.2	147.0	3706	254.5	152.0	.219
	48.5	195.0	4272	325.1	202.0	.208
	54.1	195.0	4270	318.1	202.0	.221
	61.0	193.0	4280	312.0	200.0	.230
	67.5	192.0	4307	305.3	198.0	.242
	43.2	228.0	3592	333.6	235.0	.218
	54.1	231.0	3589	327.4	236.0	.237
	61.0	229.0	3594	320.7	234.0	.250
	64.5	225.0	3595	313.5	230.0	.259
	70.8	226.0	3576	309.6	251.0	.277
Diethyl Carbitol	49.0	97.3	4054	280.8	106.1	0.141
	53.3	101.5	4138	271.7	109.1	.155
	57.5	96.5	4139	263.4	104.0	.158
	62.2	99.4	4072	255.6	106.5	.166
	67.7	97.3	4052	246.5	104.0	.172
	47.6	147.0	3972	297.1	158.0	.171
	51.8	147.0	3998	292.0	154.0	.176
	56.2	146.0	3996	288.8	151.0	.177
	60.6	146.0	3997	284.1	152.0	.183
	65.3	150.0	4006	279.3	156.0	.197
	50.5	200.0	3116	298.1	207.0	.204
	51.2	194.0	3869	317.9	203.0	.201
	55.7	195.0	3899	315.7	204.0	.206
	59.4	197.0	3937	316.7	206.0	.212
	62.8	194.0	3964	312.3	205.0	.218
Benzyl Cellosolve	47.7	100.0	2705	269.0	103.0	0.100
	57.2	102.0	2717	264.0	106.0	.105
	66.3	97.0	2698	258.0	103.0	.106
	72.5	101.0	2725	256.0	105.0	.111
	49.5	150.0	2717	276.0	156.0	.137
	58.6	149.0	2710	274.0	153.0	.137
	70.0	150.0	2717	262.0	153.0	.153
	74.0	149.0	2679	254.0	152.0	.162
	57.4	200.0	3054	295.0	203.0	.204
	66.0	195.0	3067	291.0	203.0	.207
	74.5	198.0	3095	286.0	202.0	.223
	52.5	158.0	3062	287.0	164.0	.151
	52.0	175.0	3065	290.0	180.0	.170
	52.5	198.0	3075	298.0	203.0	.197
	49.5	200.0	3095	300.0	205.0	.192
	53.0	224.0	3067	307.0	225.0	.233

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Fig. 1

Outlet pipe

Micarta end piece

Micarta jacket

Monel-metal tube

Inlet pipe

Thermocouples (10)

Packing nut

Potential lead

Copper terminal

Drilled hole filled with high-melt solder

Tube wall

Alundum bead

Detail of thermocouple installation

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Figure 1. - Heat-transfer unit.

Potential
transformer

Heat-transfer
unit

Power
transformer

Voltage
regulator

a-c. line

Current
transformer

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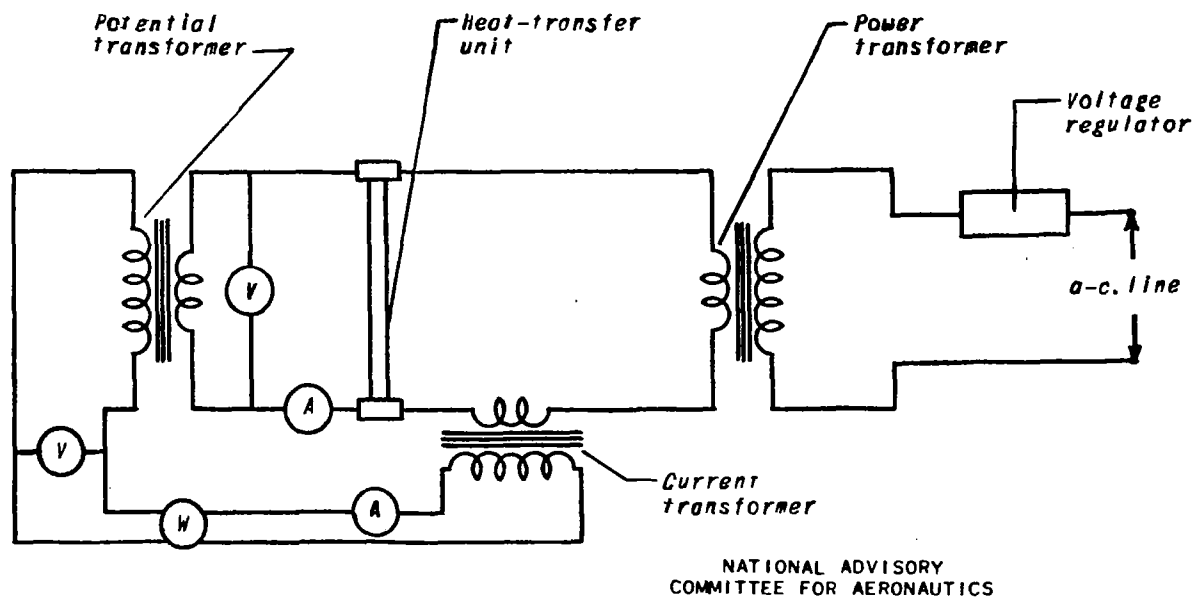
Figure 2. - Wiring diagram of heat-transfer setup.

Fig. 2
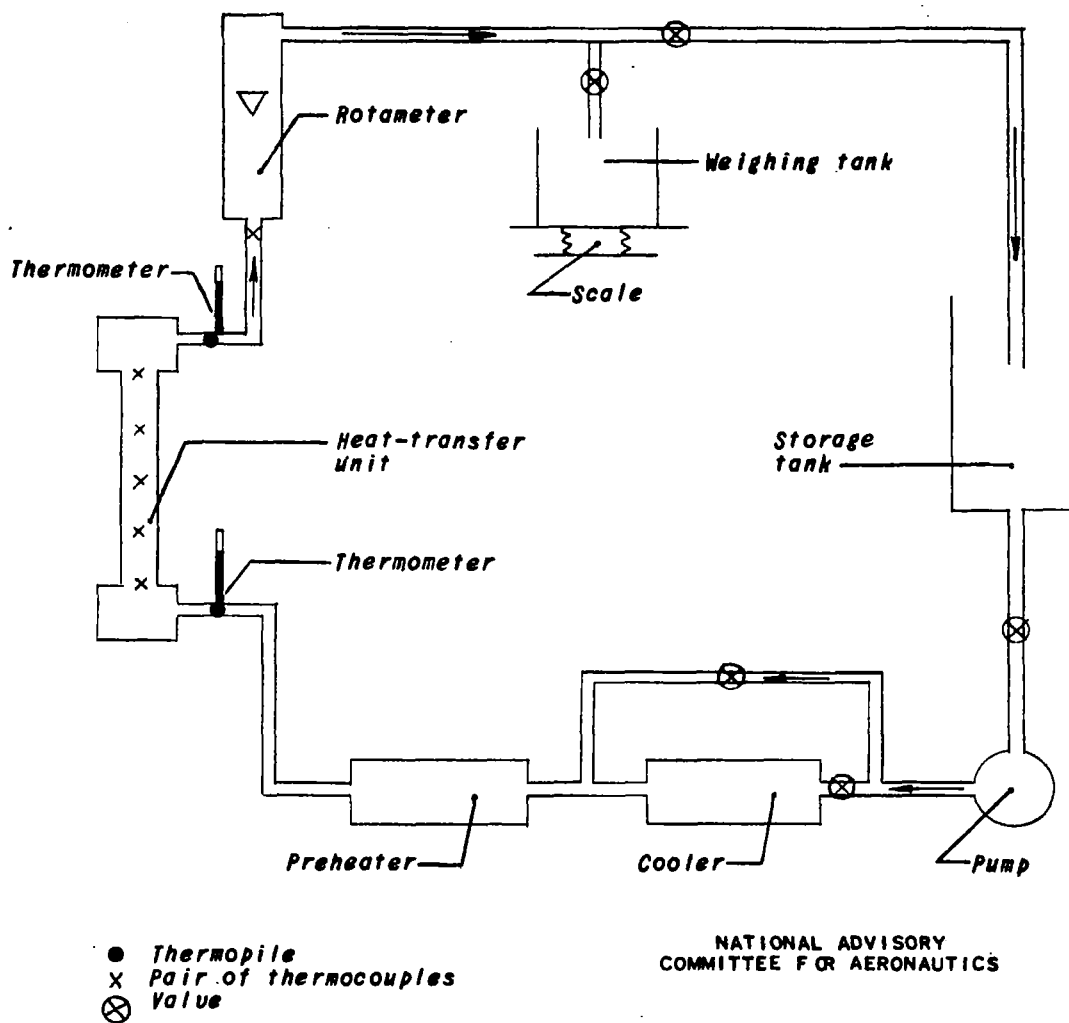
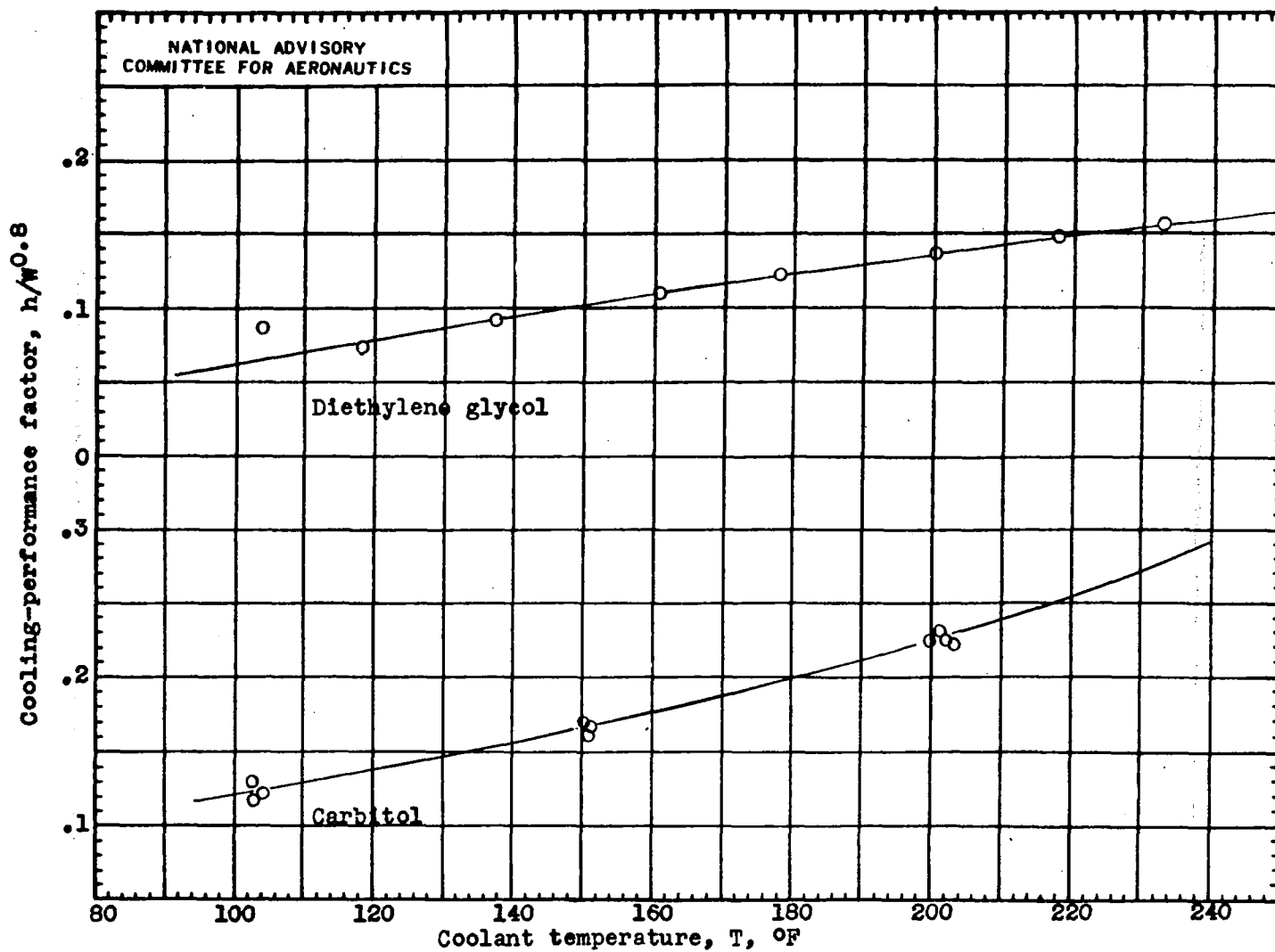Figure 3. - Coolant-flow diagram.
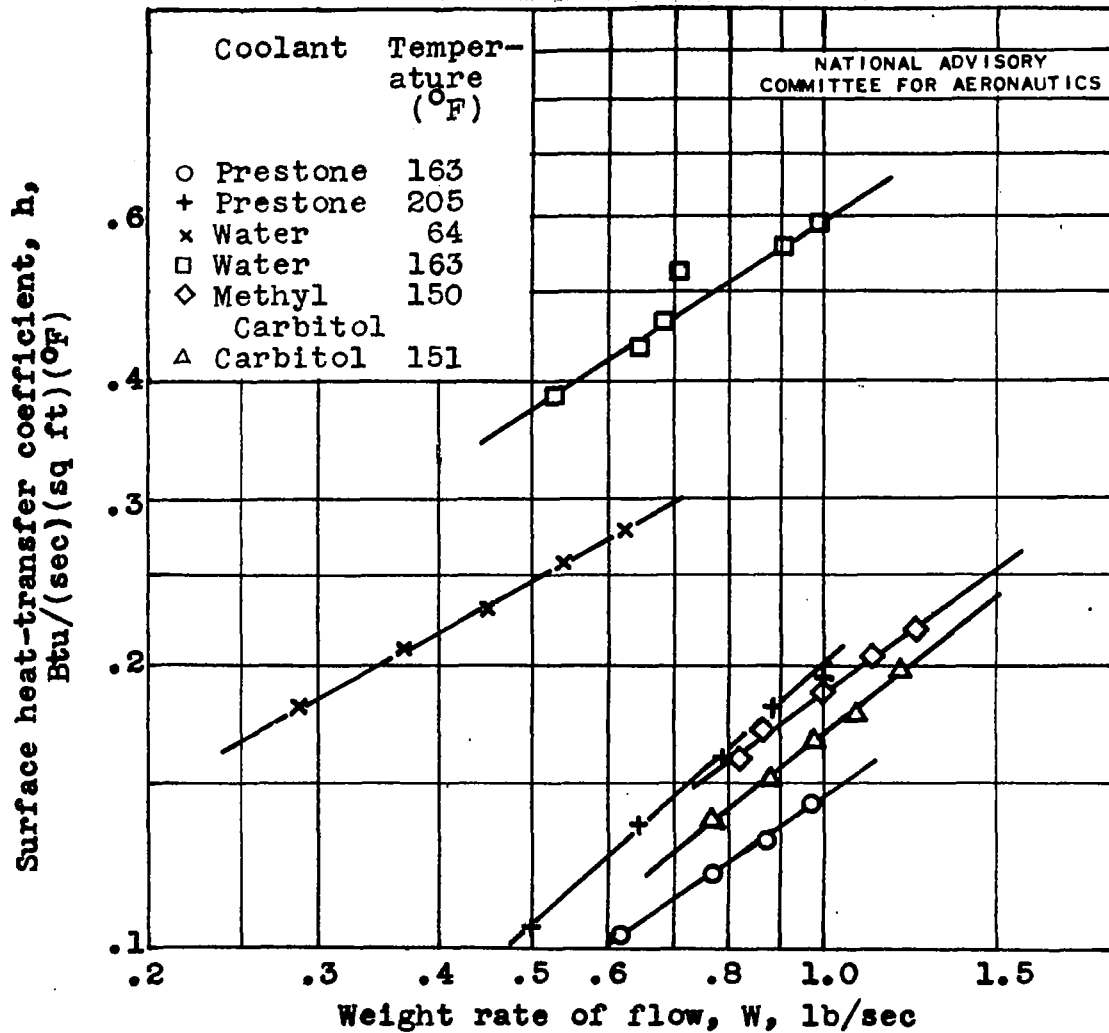
Figure 4.— Effect of weight rate of flow on the surface
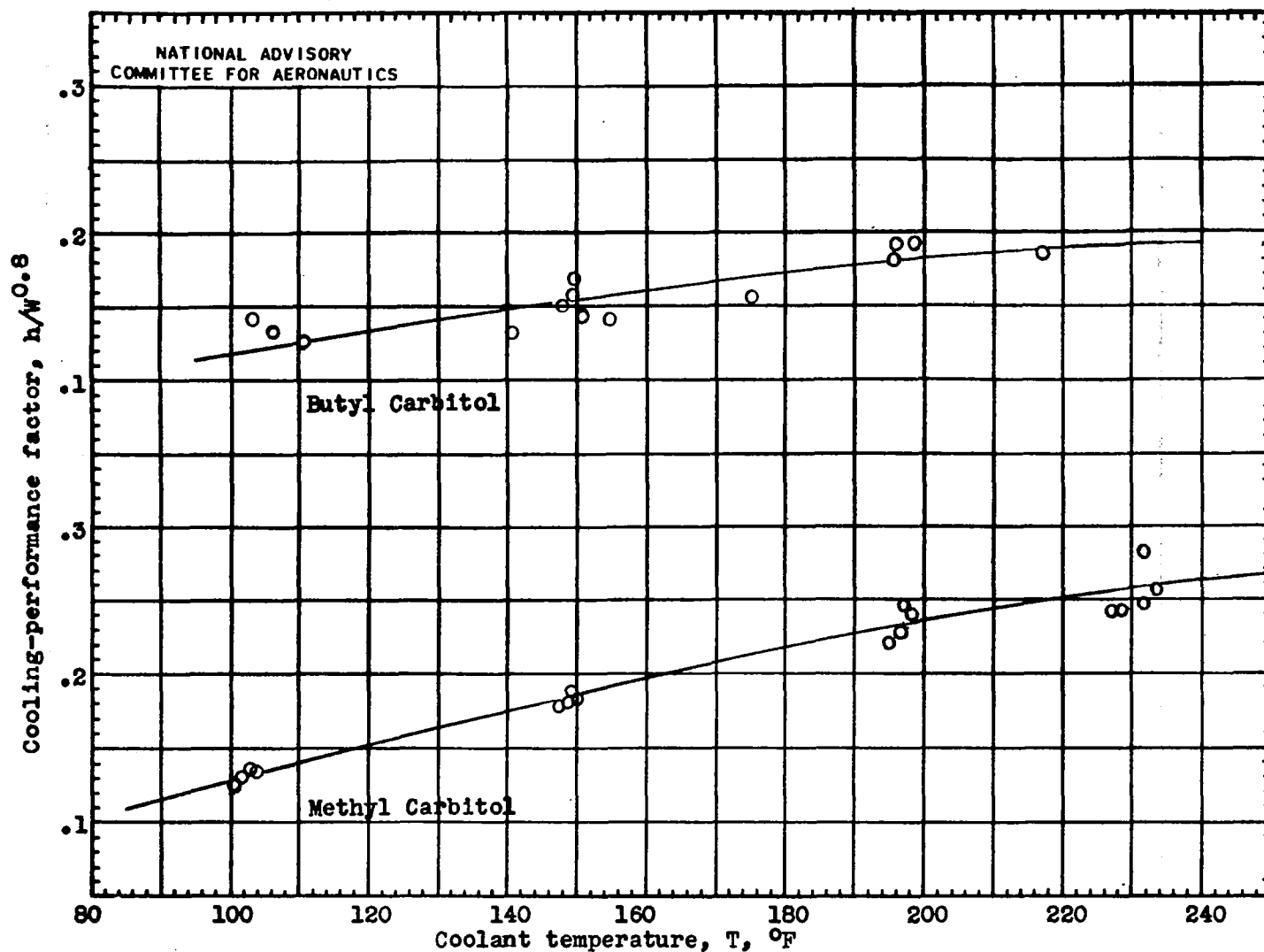   heat-transfer coefficients for several coolants.
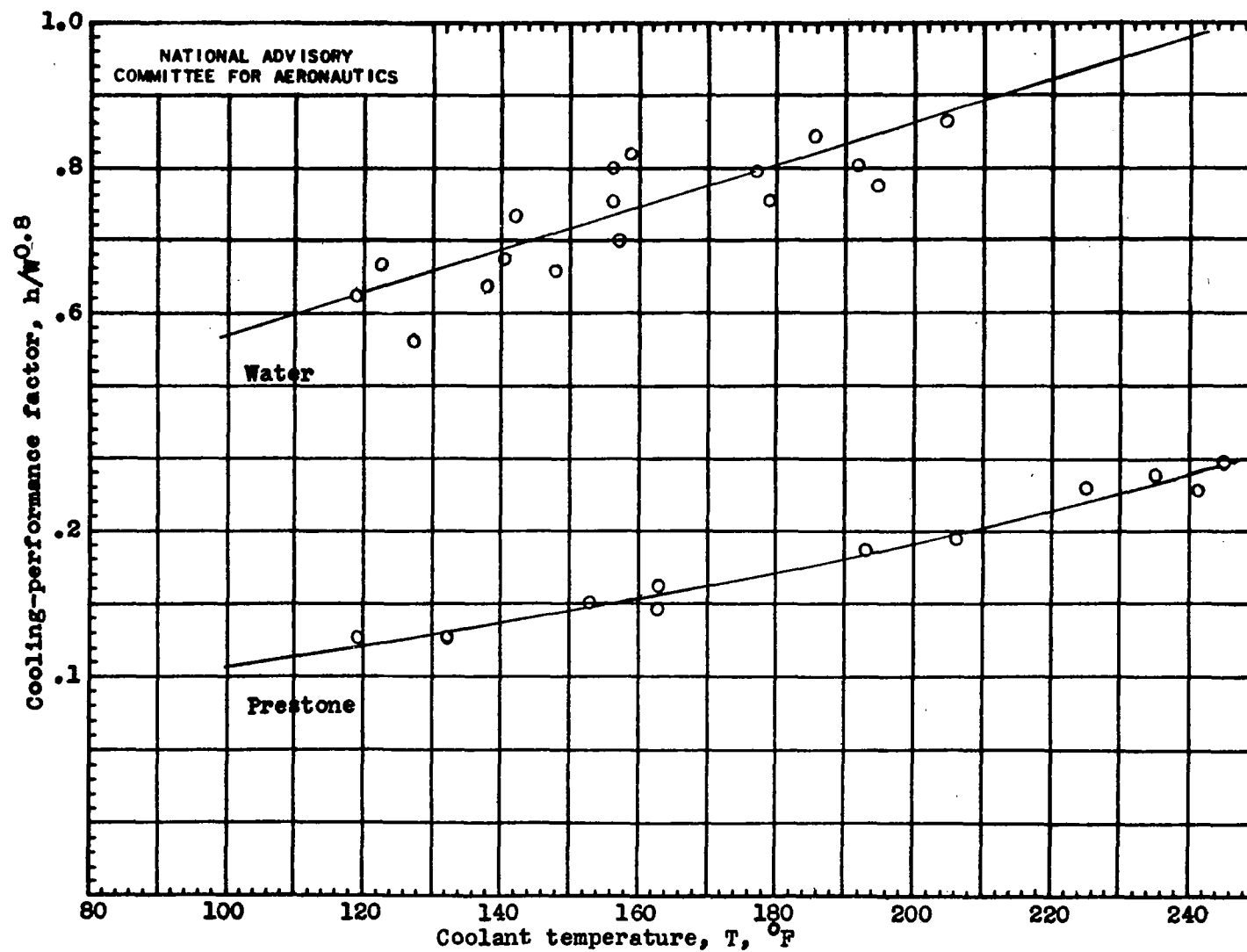
(a) Water and Prestone.

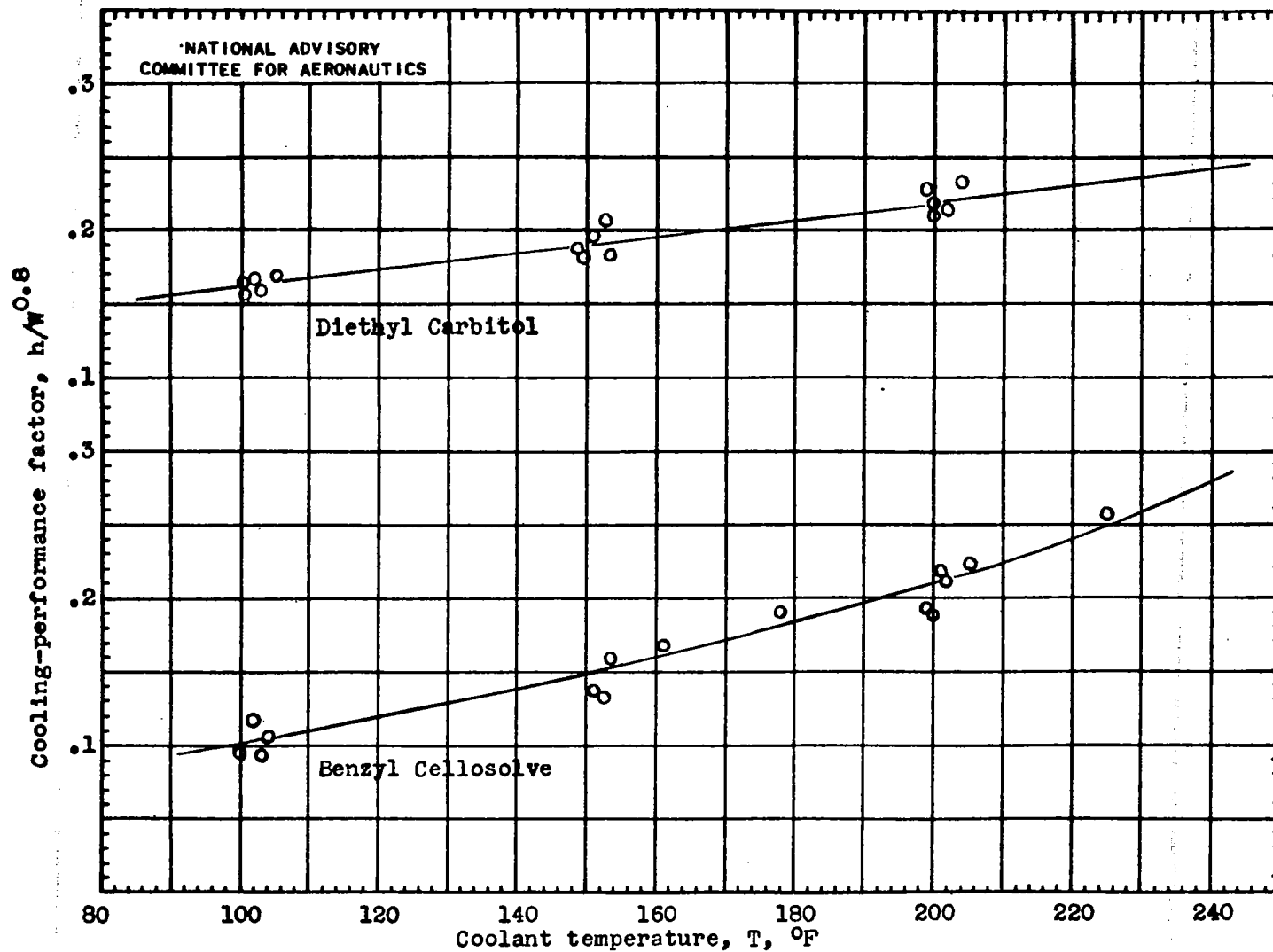Figure 5.— Variation of cooling-performance factor with coolant temperature.

(b) Diethylene glycol and Carbitol.

Figure 5.— Continued.

(c) Butyl Carbitol and Methyl Carbitol.

Figure 5.- Continued.

(d)   Diethyl Carbitol and Benzyl Cellosolve.

Figure 5.-   Concluded.